CRITICAL HEAT LOADS IN A HIGH-SPEED TWO-PHASE FLOW AT LOW PRESSURES

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A description is given of a possible model of burnout in a high-speed gas-liquid flow and of a method of calculation based on this model. An experimental apparatus is briefly described, together with the results of a study of the critical heat fluxes for a steam-water mixture in the dryness range above 0.5 at pressures to 10^6 N/m^2. The experimental data are compared with the theoretical values.

Recently, much attention has been devoted to the possibility of cooling intensely heated surfaces by means of gas-liquid and vapor-liquid mixtures with a low volume content of condensed phase.

Under these conditions heat transfer is much higher than for pure vapor cooling and even higher than for cooling with a liquid at the same mass flow rate. At high flow velocities a gas-liquid mixture moves in an annular or disperse-annular flow regime. At the surface of the tube there is formed a very thin, gas-entrained liquid film with a velocity greater than that in the wall layer for purely liquid cooling (at the same mass flow rate). The rest of the liquid is dispersed in the gaseous core of the flow. Between the film and the core there is an exchange of liquid, which intensifies heat transfer and supplies liquid to the film, thus to some extent compensating for evaporation. Moreover, the higher heat conductivity and the higher, as compared with the vapor, density of the liquid in the wall layer create much more favorable conditions for heat transfer than vapor moving at the same velocity. For these reasons both Soviet and foreign scientists are showing considerable interest in this method of cooling.

Recent studies [1–3] show that it is possible to assume the existence of two different burnout mechanisms in forced flows of gas-liquid mixtures. In a flow of subcooled liquid and in the region of low volume contents of gas phase burnout is connected with the fact that a transition from nucleate to film boiling takes place in the wall layer. At higher gas phase contents and high flow velocities burnout has a rather different character and generally occurs at lower heat loads than in ordinary boiling.

If the flow velocity of the liquid in the film is sufficiently high, the vaporization process at the wall may be suppressed by convection and liquid may be evaporated only from the surface of the film. In this case the essential difference from the burnout mechanism in ordinary boiling is that there is a sharp deterioration in heat transfer conditions as a result of the total evaporation of the liquid film. The existence of burnout of this type is confirmed by visual observations and motion-picture photography of a heated plate placed in a water-air flow moving at high velocity. The frames reproduced in Fig. 1 show how the film disappears as the heat load is increased from 0 to $1.5 \cdot 10^6$ W/m^2, whereupon the plate is destroyed (the motion-picture photography was carried out at the Thermophysics Department of the Kalinin Leningrad Polytechnic Institute by M. E. Lavrent'ev).

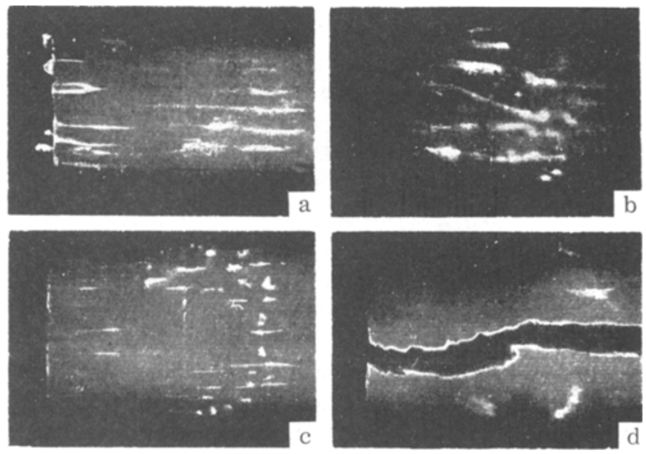

Fig. 1. Burnout of a plate in a disperse-annular water-air flow (direction of flow from right to left): a) at $q = 0$ (continuous film breaking up at the edge), b) $q \neq 0$ (edge of continuous film recedes from edges of plate), c) $q = 1.5 \cdot 10^6$ W/m^2, d) destruction of plate.

In the disperse-annular flow regime the wall layer of liquid is continually replenished with droplets from the flow core. Therefore the critical heat flux must depend both on the flowrate of liquid in the film and on the rate of replenishment.

At present, most effort is being concentrated on creating a burnout model for a disperse-annular flow that is amenable to mathematical description [2–4]. The construction of a sufficiently reliable calculation method requires a detailed knowledge of the process of diffusion of the liquid droplets, the distribution of liquid over the cross section, and the stability conditions for the film. These questions have still not been adequately investigated. In general, attempts to calculate the critical conditions have been associated with the introduction of a number of coefficients, unknown in the general case, which have to be determined experimentally [1–3].

The results of comparing experimental data with the values calculated from the recommended expressions are often unsatisfactory. Moreover, almost all the published data relate to the high-pressure region.

In this paper we propose a model which can be used to calculate the critical heat flux in the disperse-annular flow regime at low pressures.

It is assumed that for zero heat flux the wall film has constant thickness. The flows of liquid to and from the film are equal. If we supply an amount of heat to the tube wall greater than that required to evaporate the liquid supplied from the flow core, then the film will grow thinner. It is assumed that there is no vaporization at the tube wall and liquid is evaporated only from the film surface. At high, near-critical heat loads, even at a small distance from the inlet the film becomes thin enough for stripping of liquid from its surface to become negligible and all the moisture from the core is trapped by the film. The total disappearance of liquid from the wall leads to a sharp deterioration in heat transfer conditions (burnout), since evaporation of the liquid from the flow core is insufficient to remove the heat supplied to the wall. In [5] it was shown that the flux of solid particles precipitated from a turbulent gas flow can be approximately calculated from the expression

$$G_c = \frac{\varepsilon}{8} c w_g = \frac{A}{8} \mathrm{Re}_g^n c w_g. \qquad (1)$$

The present authors have made an additional study of the precipitation of liquid droplets from the disperse core of a high-speed flow. It was found that in the region of mass moisture contents from 0.1 to 0.5 the coefficient A depends only slightly on the concentration of liquid in the flow core. In this range of moisture contents in the first approximation for fine droplets we may take $A = 0.316$, as in the usual Blasius expression for the drag coefficient in a single-phase flow. In exactly the same way, the value $n = -0.25$ proves to be correct.

The balance equation for the liquid dispersed in the core for an element of tube length, neglecting the thickness of the film as compared with the tube diameter, is written in the form

$$\pi D G_c \, dz + \frac{\pi D^2}{4} w_g dc = 0. \qquad (2)$$

For a relatively small change in vapor content along the tube it is possible to assume that the velocity is constant along the length. Then integration of (2) using (1) with boundary condition $c = c_0$, when $z = 0$, gives an expression for the change in the concentration, averaged over the cross section, of liquid in the core along the length of the tube:

$$c = c_0 \exp\left[-\frac{A}{2} \mathrm{Re}_g^n \frac{z}{D}\right]. \qquad (3)$$

With our initial assumptions the balance equation for the liquid in the film for a tube element may be written in the following form:

$$\frac{\pi D q_{cr} dz}{\Delta i} + dG_f - G_c \, \pi D dz = 0. \qquad (4)$$

The latter equation is integrated with the boundary conditions

$$z = 0, \quad G_f = G_{f\,0},$$
$$z = z_{cr}, \quad G_f = 0. \qquad (5)$$

As a result, for the specific critical flux we get

$$q_{cr} = \frac{\Delta i}{\pi D z_{cr}} \left\{ G_{f\,0} + G_{c\,0} \left[1 - \exp\left(-\frac{A \mathrm{Re}_g^n z_{cr}}{2D}\right)\right]\right\}. \qquad (6)$$

For very short tubes Eq. (6) does not hold, since as the heat load increases conditions develop under which boiling at the wall becomes inevitable and burnout is associated with instability of the two-phase layer at the wall.

Equation (6) makes it possible to calculate the critical heat flux if the distribution of liquid between the film and the disperse core is known at the inlet to the heated channel. On the basis of an analysis of quite extensive data obtained by various authors [6–10] on the liquid flow rates in the film in water-air flows we obtained the equation

$$\frac{G_f}{G_l} = 0.985 - 0.44 \log\left[\frac{\rho}{\rho_l}\left(\frac{\mu_l w_g}{\sigma}\right)^2 \cdot 10^4\right]. \qquad (7)$$

Equation (7) describes all the experimental data of [6–10] with an accuracy of ±30%. In this case the parameter $(\rho_g/\rho_l)(\mu_l w_g/\sigma)^2$ was varied within the range $(1-100) \cdot 10^{-4}$, which in all cases corresponded to a disperse-annular flow regime of the gas-liquid mixture. It should be noted that (7) was obtained on the basis of experimental data on water-air flows only, since there is no such information in the literature concerning vapor-liquid mixtures.

Equation (7) is easily solved for G_f/G_l by the method of successive approximations.

The proposed model will be incorrect in the case of vaporization at the wall, since this may lead to loss of stability and stripping of the film even before its total evaporation.

Below we present the results of an experimental study of burnout in the disperse-annular flow regime for a steam-water mixture. The region of pressures covered extended from 1.96 to $9.8 \cdot 10^5$ N/m^2. The experiments were conducted on stainless steel tubes $5 \cdot 10^{-3}$ m in internal diameter and of length $L = 0.240$ and 0.425 m. Dry steam from a boiler was fed into the inlet of the apparatus. In a mixer water was injected into the steam and the steam-water mixture obtained was fed into the working section, and then into a condenser. The injected water was preheated in an electric furnace.

The mass vapor content was varied in the range from 0.5 to 0.9. The investigated range of vapor mass flow rates was 200–1200 kg/m^2·sec, which corresponds to a liquid content by volume of less than 1%.

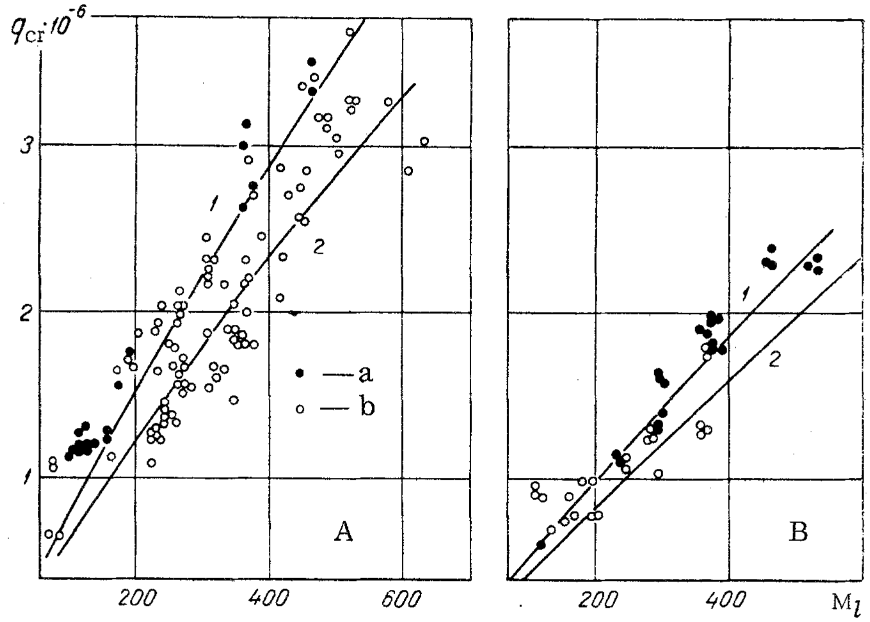

Fig. 2. Specific critical heat flux q_{cr} (W/m^2) as a function of the total mass flow rate of water M_l ($kg/m^2 \cdot sec$) in the flow for $P = 2.94 \cdot 10^5$ N/m^2 and $L = 0.24$ (A) and 0.425 (B) m: 1 and a) by calculation and from experimental data for $M_g = 318$ $kg/m^2 \cdot sec$; 2 and b) the same for $M_g = 500$ $kg/m^2 \cdot sec$.

In this interval the maximum values of the vapor mass velocity were reached only at the highest pressures. The vapor flow rate was determined from the measured volume of condensate formed, the known water injection rate, and the parameters of the steam and the water before mixing. As a check, the heat balance was computed. Over the entire test interval we noted pressure pulsations with an amplitude of the order of 10^4 N/m^2 and a period of about 0.5 sec. Since the pressure pulsations are accompanied by flow rate pulsations, while the vapor flow rate was not continuously recorded, in practice we determined the average vapor flow rate. The maximum error was in determining the injection flow rate. In the region of small flow rates the relative measuring error reached 5%.

The working section was heated with alternating current. The power output was computed from measured values of the voltage drop on the working section and the current. The maximum measuring error did not exceed 1%.

To record burnout, a thermocouple, connected to the input of a regulating electronic potentiometer, was soldered to the wall 10 mm from the outlet end of the working section. When the wall temperature at the site of the thermocouple reached 900° K the potentiometer relay automatically disconnected the electrical supply to the working section. To reach critical conditions, the power supplied was continuously increased until stable temperature fluctuations appeared at the anticipated burnout site. The amplitude of these fluctuations sometimes reached tens of degrees. The value of the heat flux corresponding to this instant was taken as the critical value, since a further slight increase in the power supplied led instantaneously to a sharp increase in temperature at the burnout site. The existence of stable wall temperature fluctuations preceding burnout has also been noted by other authors. It may be connected with the presence of certain flow rate pulsations in the system.

The results of the experiments are presented in Fig. 2, for two vapor flow rates as parameter. The figure gives the dependence of the specific critical heat flux on the total amount of liquid in the flow. The continuous lines correspond to the values calculated from Eq. (6). It was assumed that $A = 0.3164$ and $n = -0.25$, while the Re_g number was calculated from the formula

$$Re_g = 4G_g/\pi D \mu_g. \tag{8}$$

In the range of parameters investigated the pressure drop along the length of the working section was of the same order as the average pressure determined as the arithmetic mean of the pressures at the inlet and outlet of the working section. This created additional difficulties in analyzing the experimental data, since the flow parameters varied considerably along the length.

It is clear from the figure that q_{cr} increases with increase in M_l and falls with increase in vapor flow rate. Comparison of Figs. 2A and 2B reveals that an increase in the length of the section leads to a decrease in q_{cr}.

Comparison of the experimental and calculated results shows that Eq. (6) gives values of q_{cr} close to the experimental values.

NOTATION

G_c—specific mass flux of droplets from the core; ε—drag coefficient, c-concentration of liquid in flow core, kg/m^2; w—average flow rate; Re—Reynolds number; A, n—constants; D—tube diameter; z—longitudinal coordinate; q—specific heat flux; G—total flow rate; M—mass velocity; Δi—change of enthalpy upon vaporization; μ—dynamic viscosity; ρ—density; σ—surface tension; $\bar{\rho} = \rho_g[1 + G_l(1 - G_f/G_l)/F\rho_g w_g]$; F—tube cross-sectional area; P—average pressure. Subscripts: g—gas phase; l—liquid phase; f—film; sus—liquid suspended in gaseous flow core; o-inlet section of heated length; cr—critical conditions.

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